

Cryogenic vessels — Static vacuum insulated vessels —

Part 2: Design, fabrication, inspection and testing

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National foreword

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Réceptacles cryogéniques - Réceptacles fixes isolés sous vide
- Partie 2: Conception, fabrication, inspection et essais

Kryo-Behälter - Ortsfeste vakuum-isolierte Behälter - Teil 2:
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This European Standard was approved by CEN on 12 August 2002.

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Foreword

This document (EN 13458-2:2002) has been prepared by Technical Committee CEN/TC 268 "Cryogenic vessels", the secretariat of which is held by AFNOR.

This European Standard shall be given the status of a national standard, either by publication of an identical text or by endorsement, at the latest by May 2003, and conflicting national standards shall be withdrawn at the latest by May 2003.

This document has been prepared under a mandate given to CEN by the European Commission and the European Free Trade Association, and supports essential requirements of EU Directive(s).

For relationship with EU Directive(s), see informative Annex ZA, which is an integral part of this document.

In this European Standard the annexes A, B E, G, I and K are normative and the annexes C, D, F, H and J are informative.

EN 13458 consists of the following Parts under the general title, *Cryogenic vessels – Static vacuum insulated vessels*

- *Part 1: Fundamental requirements*
- *Part 2: Design, fabrication, inspection and testing*
- *Part 3: Operational requirements*

According to the CEN/CENELEC Internal Regulations, the national standards organizations of the following countries are bound to implement this European Standard: Austria, Belgium, Czech Republic, Denmark, Finland, France, Germany, Greece, Iceland, Ireland, Italy, Luxembourg, Malta, Netherlands, Norway, Portugal, Spain, Sweden, Switzerland and the United Kingdom.

1 Scope

This European Standard specifies requirements for the design, fabrication, inspection and testing of static vacuum insulated cryogenic vessels designed for a maximum allowable pressure of more than 0,5 bar.

This European Standard is applicable to static vacuum insulated cryogenic vessels for fluids as specified in EN 13458-1, and does not apply to vessels designed for toxic fluids.

For static vacuum insulated cryogenic vessels designed for a maximum allowable pressure of not more than 0,5 bar this standard can be used as a guide.

2 Normative references

This European Standard incorporates by dated or undated reference, provisions from other publications. These normative references are cited at the appropriate places in the text, and the publications are listed hereafter. For dated references, subsequent amendments to or revisions of any of these publications apply to this European Standard only when incorporated in it by amendment or revision. For undated references the latest edition of the publication referred to applies (including amendments).

EN 287-1, *Approval testing of welders – Fusion welding – Part 1: Steels.*

EN 287-2, *Approval testing of welders – Fusion welding – Part 2: Aluminium and aluminium alloys.*

EN 288-3:1992, *Specification and approval of welding procedures for metallic materials – Part 3: Welding procedure tests for the arc welding of steels.*

EN 288-4:1992, *Specification and approval of welding procedures for metallic materials – Part 4: Welding procedure tests for the arc welding of aluminium and its alloys.*

EN 288-8, *Specification and approval of welding procedures for metallic materials – Part 8: Approval by a pre-production welding test.*

EN 473, *Qualification and certification of NDT personnel – General principles.*

EN 875:1995, *Destructive tests on welds in metallic materials – Impact tests – Test specimen location, notch orientation and examination.*

EN 895:1995, *Destructive tests on welds in metallic materials – Transverse tensile test.*

EN 910:1996, *Destructive tests on welds in metallic materials – Bend tests.*

EN 1252-1:1998, *Cryogenic vessels – Materials – Part 1: Toughness requirements for temperatures below –80 °C.*

EN 1252-2, *Cryogenic vessels – Materials – Part 2: Toughness requirements for temperatures between –80 °C and –20 °C.*

EN 1418, *Welding personnel – Approval testing of welding operators for fusion welding and resistance weld setters for fully mechanised and automatic welding of metallic materials.*

EN 1435, *Non-destructive examination of welds – Radiographic examination of welded joints.*

EN 1626, *Cryogenic vessels – Valves for cryogenic service.*

EN 1797, *Cryogenic vessels – Gas/material compatibility.*

EN 10028-4, *Flat products made of steels for pressure purposes – Part 4: Nickel alloy steels with specified low temperature properties.*

EN 10028-7:2000, *Flat products made of steels for pressure purposes – Part 7: Stainless steels.*

prEN 10216-5, *Seamless steel tubes for pressure purposes - Technical delivery conditions - Part 5: Stainless steel tubes.*

prEN 10217-7, *Welded steel tubes for pressure purposes - Technical delivery conditions - Part 7: Stainless steel tubes.*

EN 12300, *Cryogenic vessels – Cleanliness for cryogenic service.*

EN 13068-3, *Non-destructive testing – Radioscopic testing – Part 3: General principles of radioscopic testing of metallic materials by X- and gamma rays.*

EN 13133, *Brazing – Brazer approval.*

EN 13134, *Brazing – Procedure approval.*

EN 13445-3, *Unfired pressure vessels – Part 3: Design.*

EN 13445-4, *Unfired pressure vessels – Part 4: Fabrication.*

EN 13458-1:2002, *Cryogenic vessels – Static vacuum insulated vessels - Part 1: Fundamental requirements.*

prEN 13458-3, *Cryogenic vessels – Static vacuum insulated vessels – Part 3: Operational requirements.*

prEN 13648-1, *Cryogenic vessels – Safety devices for protection against excessive pressure – Part 1: Fundamental requirements*

prEN 13648-3, *Cryogenic vessels – Safety devices for protection against excessive pressure – Part 3: Determination of required discharge – Capacity and sizing.*

prEN ISO 4126-2, *Safety devices for protection against excessive pressure – Part 2: Bursting disc safety devices (ISO/DIS 4126-2:1996).*

EN ISO 6520-1:1998, *Welding and allied processes – Classification of geometrical imperfections in metallic materials – Part 1: Fusion welding (ISO 6520-1:1998).*

ISO 1106-1, *Recommended practice for radiographic examination of fusion welded joints - Part 1: Fusion welded butt joints in steel plates up to 50 mm thick.*

SA-353/A353M, *Specification for pressure vessel plates, alloy steel, 9 percent nickel, double-normalized and tempered.*

SA-479/SA-479M, *Specification for stainless steel bars and shapes for use in boilers and other pressure vessels.*

SA-522/SA-522M, *Specification for forged or rolled 8 and 9% nickel alloy steel flanges, fittings, valves and parts for low-temperature service.*

SA-553/SA-553M, *Specification for pressure vessel plates, alloy steel quenched and tempered 8 and 9 percent nickel.*

3 Terms, definitions and symbols

3.1 Terms and definitions

For the purposes of this European Standard, the following terms and definitions apply.

3.1.1

static vessel

stationary unit capable of receiving, storing (under pressure) and dispensing cryogenic fluids. The vessel is not intended to be used for transporting liquid product

3.1.2

inner vessel

pressure vessel proper intended to contain the cryogenic fluid

3.1.3

outer jacket

gas-tight enclosure which contains the inner vessel and enables a vacuum to be established

3.1.4

automatic welding

welding in which the parameters are automatically controlled. Some of these parameters may be adjusted to a limited extent, either manually or automatically, during welding to maintain the specified welding conditions

3.1.5

maximum allowable pressure, p_s

maximum pressure for which the equipment is designed, as specified by the manufacturer, defined at a location specified by the manufacturer, being the location of connection of protective or limiting devices or the top of the equipment

NOTE p_s is equivalent to PS used in article 1, 2.3 of the PED.

3.1.6

relief plate/plug

plate or plug retained by atmospheric pressure only which allows relief of excess internal pressure

3.1.7

bursting disc device

non-reclosing pressure relief device ruptured by differential pressure. It is the complete assembly of installed components including where appropriate the bursting disc holder

3.2 Symbols

NOTE Throughout this European Standard p_s is equivalent to PS used in article 1, 2.3 of the PED and p_T is equivalent to PT used in Annex I of the PED.

For the purposes of this standard, the following symbols apply:

c	allowances	mm
d_i	diameter of opening	mm
d_a	outside diameter of tube or nozzle	mm
f	narrow side of rectangular or elliptical plate	mm
l_b	buckling length	mm

n	number	-
p	design pressure as defined by 4.2.3.2 j) and 4.3.3.2	bar
p_e	allowable external pressure limited by elastic buckling	bar
p_k	strengthening pressure	bar
p_p	allowable external pressure limited by plastic deformation	bar
p_T	pressure test (see 4.2.3.2 g))	bar
r	radius e.g. inside knuckle radius of dished end and cones	mm
s	minimum wall thickness	mm
s_e	actual wall thickness	mm
v	factor indicative of the utilisation of the permissible design stress in joints or factor allowing for weakenings	-
x	(decay-length zone) distance over which governing stress is assumed to act	mm
A	area	mm ²
A_5	elongation at fracture	
C	design factors	-
D	shell diameter	mm
D_a	outside diameter e.g. of a cylindrical shell	mm
D_i	internal diameter e.g. of a cylindrical shell	mm
E	Young's modulus	N/mm ²
I	moment of inertia of stiffening ring	mm ⁴
K	material property (see 4.3.2.3.1)	N/mm ²
K_{20}	see 4.3.2.3.2	
K_t	see 4.3.2.3.3	
K_{design}	a value defined by the manufacturer for a particular design case	
R	radius of curvature e.g. inside crown radius of dished end	mm
S	safety factor at design pressure	-
S_k	safety factor against elastic buckling at design pressure	-
S_p	safety factor against plastic deformation at design pressure	-
S_T	safety factor against plastic deformation at proof test pressure	-

Z	auxiliary value	-
ν	Poisson's ratio	-
u	out of roundness	
σ_k	design stress value	N/mm ²

4 Design

4.1 Design options

4.1.1 General

The design shall be carried out in accordance with one of the options given in 4.1.2, 4.1.3 or 4.1.4.

In the case of 9 % Ni steel, the additional requirements of annex B shall be satisfied.

For carbon and low alloy steels the requirements of EN 1252-2 shall be satisfied.

When further use of cold properties is considered the requirements of annex E shall be satisfied.

4.1.2 Design by calculation

Calculation of all pressure and load bearing components shall be carried out. The pressure part thicknesses of the inner vessel and outer jacket shall not be less than required by 4.3. Additional calculations may be required to ensure the design is satisfactory for the operating conditions including an allowance for external loads (e.g. seismic).

4.1.3 Design by calculation when adopting pressure strengthening

The pressure retaining capability of inner vessels manufactured from austenitic stainless steel, strengthened by pressure, is calculated in accordance with the informative annex C.

4.1.4 Design by calculation supplemented with experimental methods

Where it is not possible to design by calculation alone planned and controlled experimental means may be used providing that the results confirm the standards of design required in 4.3. An example would be the application of strain gauges to assess stress levels.

4.2 Common design requirements

4.2.1 General

The requirements of 4.2.2 to 4.2.8 are applicable to all vessels irrespective of the design option used.

In the event of an increase in at least one of the following parameters:

- maximum allowable pressure;
- specific mass (density) of the densest gas for which the vessel is designed;
- maximum tare weight of the inner vessel;
- nominal length and/or diameter of the inner shell;

or, in the event of any change relative:

- to the type of material or grade (e.g. stainless steel to aluminium or change of stainless steel grade);
- to the fundamental shape;
- to the decrease in the minimum properties of material being used;
- to the modification of the design of an assembly method concerning any part under stress, particularly as far as the support systems between the inner vessel and the outer jacket or the inner vessel itself or the protective frame, if any, are concerned;

the initial design programme shall be repeated to take account of these modifications.

4.2.2 Design specification and documentation

To enable the design to be prepared the following information shall be available:

- maximum allowable pressure;
- fluids intended to be used;
- liquid capacity;
- volume of the inner vessel;
- configuration;
- method of handling and securing during transit and site erection;
- site conditions e.g. ambient temperatures, seismic etc.;
- fill and withdrawal rates.

A design document in the form of drawings with text if any shall be prepared, it shall contain the information given above plus the following where applicable:

- definition of which components are designed by calculation, by pressure strengthening, by experiment and by satisfactory in-service experience;
- drawings with dimensions and thicknesses of load bearing components;
- specification of all load bearing materials including grade, class, temper, testing etc. as relevant;
- type of material test certificates;
- location and details of welds and other joints, welding and other joining procedures, filler, joining materials etc. as relevant;
- calculations to verify compliance with this standard;
- design test programme;
- non destructive testing requirements;
- pressure test requirements;
- piping configuration including type, size and location of all valves and relief devices;

- details of lifting points and lifting procedure;
- wind, seismic loads.

4.2.3 Design loads

4.2.3.1 General

Static vessels are not considered to be in cyclic service, therefore fatigue analysis needs normally not to be performed.

The static cryogenic vessel shall be able to safely withstand the mechanical and thermal loads encountered during normal operation and pressure test, as specified in 4.2.3.2 to 4.2.3.7.

4.2.3.2 Inner vessel

The following loads shall be considered to act in the combinations specified in 4.2.3.2 j):

- a) pressure during operation when the vessel contains cryogenic liquid product

$$p_{cL} = p_s + p_L + 1 \text{ bar}$$

where

p_s maximum allowable pressure (in bar);

p_L pressure (in bar) exerted by the weight of the liquid contents when the vessel is filled to capacity with either:

- 1) boiling liquid at atmospheric pressure; or
- 2) cryogenic fluid at its equilibrium triple point or melting point temperature at atmospheric pressure.

p_L may be neglected if less than 5 % of $(p_s + 1)$. Otherwise the pressure in excess of 5 % of $(p_s + 1)$ shall be used;

- b) pressure during operation when the vessel contains only gaseous product at 20 °C

$$p_{cG} = p_s + 1 \text{ bar}$$

- c) reactions at the support points of the inner vessel during operation when the vessel contains cryogenic liquid product. The reactions shall be determined by the weight of the inner vessel, the weight of the maximum contents of the cryogenic liquid and vapour and seismic loadings where appropriate. The seismic loadings shall consider any forces exerted on the vessel by the insulation;
- d) reactions at the support points of the inner vessel during operation when the vessel contains only gaseous product at 20 °C. The reactions shall be determined by the weight of the inner vessel, its contents and seismic loadings where appropriate. The seismic loadings shall consider any forces exerted on the vessel by the insulation;
- e) load imposed by the piping due to the differential thermal movement of the inner vessel, the piping and the outer jacket.

The following cases shall be considered:

- cool down (inner vessel warm - piping cold);
- filling and withdrawal (inner vessel cold - piping cold); and

— storage (inner vessel cold - piping warm);

f) load imposed on the inner vessel at its support points when cooling from ambient to operating temperature.

g) pressure test : the value used for design purposes shall be the higher of:

$$p_T = 1,43 (p_s + 1) \text{ or see 6.5.1 or}$$

$$p_T = 1,25 (p_s + p_L + 1) \frac{K_{20}}{K_{design}} \text{bar}$$

considered for each element of the vessel e.g. shell, courses, head, etc.

The 1 bar is added to allow for the external vacuum;

h) loads imposed during transit and site erection;

i) load imposed by pressure in annular space equal to the set pressure of the outer jacket relief device and atmospheric pressure in inner vessel;

j) the vessel shall be capable of withstanding the following combinations of loadings. The design pressure p is equal to pressure specified therein, in each combination 1, 2 and 3:

1) operation at maximum allowable working pressure when vessel is filled with cryogenic liquid: a) + c) + e) + f);

2) operation at maximum allowable working pressure when vessel is filled with warm gas: b) + d);

3) pressure test: g);

4) shipping and lifting: h);

5) vessel subject to external pressure developed in the vacuum jacket: i).

The inner vessel shall, in addition, be capable of holding the pressure test fluid without gross plastic deformation.

4.2.3.3 Outer jacket

The following loads shall be considered to act in combination where relevant:

a) an external pressure of 1 bar;

b) an internal pressure equal to the set pressure of the outer jacket pressure relief device;

c) load imposed by the supporting systems in the outer jacket taking into consideration site conditions, e.g. wind and seismic loadings etc.;

d) load imposed by piping as defined in 4.2.3.2 e);

e) load imposed at the inner vessel support points in the outer jacket when the inner vessel cools from ambient to operating temperature and during operation;

f) loads imposed during transit and site erection;

g) external loads from e.g. wind, seismic or other site conditions;

h) gross mass.

4.2.3.4 Inner vessel supports

The inner vessel supports shall be suitable for the load defined in 4.2.3.2 c) plus loads due to differential thermal movements.

4.2.3.5 Outer jacket supports

The outer jacket supports shall be suitable for the load defined in 4.2.3.3.

4.2.3.6 Lifting points

Lifting points shall be suitable for lifting the static cryogenic vessel when empty and lifted in accordance with the specified procedure.

4.2.3.7 Piping and accessories

Piping including valves, fittings and supports shall be designed for the following loads. With the exception a) the loads shall be considered to act in combination where relevant:

- a) pressure test : in accordance with 6.5.4;
- b) pressure during operation : not less than the set pressure of the system pressure relief devices, e.g. set pressure of the thermal relief device;
- c) thermal loads defined in 4.2.3.2 e);
- d) loads generated during pressure relief discharge;
- e) a design pressure not less than the maximum allowable pressure p_s of the inner vessel plus any appropriate liquid head. For piping inside the vacuum jacket a further 1 bar shall be added.

4.2.4 Corrosion allowance

Corrosion allowance is not required on surfaces in contact with the operating fluid. Corrosion allowance is not required on other surfaces if they are adequately protected against corrosion.

4.2.5 Inspection openings

Inspection openings are not required in the inner vessel or the outer jacket, providing the requirements of prEN 13458-3 are followed.

NOTE 1 Due to the combination of materials of construction and operating fluids, internal corrosion cannot occur.

NOTE 2 The inner vessel is inside the evacuated outer jacket and hence external corrosion of the inner vessel cannot occur.

NOTE 3 The elimination of inspection openings also assists in maintaining the integrity of the vacuum in the interspace.

4.2.6 Pressure relief

4.2.6.1 General

Relief devices for the inner vessel shall be in accordance with prEN 13648-1.

Relief devices for the outer jacket shall be in accordance with annex I.

4.2.6.2 Inner vessel

The inner vessel shall be provided with a pressure limiting system to protect the vessel against excessive pressure. Examples of current practice are shown in annex D. The system shall:

- be designed so that it is fit for purpose;
- be independent of other functions, unless its safety function is not affected by such other functions;
- limit the vessel pressure to 110 % maximum allowable pressure in a momentary surge;
- fail safely;
- contain redundant features;
- contain non-common mode failure mechanisms (diversity).

The capacity of the protection system shall be established by considering all of the probable conditions contributing towards internal excess pressure. For example:

- a) normal vessel heat leak;
- b) heat leak with loss of vacuum;
- c) failure in the open position of the make-up pressure control valve;
- d) any other valve in a line connecting a high pressure source to the inner vessel;
- e) recycling of any possible combination of pumps;
- f) flash gas, plus liquid, from maximum plant capacity fed into a tank which is at operating temperature.

The excess pressure created by any combination of conditions 'a' to 'f' shall be limited to not more than 110 % of maximum allowable pressure by at least one re-closable device. The required capacity of this re-closable device may be calculated in accordance with prEN 13648-3.

NOTE Where, in addition, a non re-closable, fail open device is fitted, its operating pressure should be chosen such that its ability to retain pressure is unaffected by the operation of the re-closable device at 110 % of maximum allowable pressure and is, in any case, not more than the top of vessel strength test pressure less 1 bar. The required capacity of any device provided for redundancy should be equal to the required capacity of the primary device.

An external fire condition only to be considered if determined by location of the cryogenic vessel.

Shut off valves or equivalent may be installed upstream of pressure relief devices, provided that interlocks are fitted to ensure that the vessel has sufficient relief capacity at all times.

The relief valve system piping shall be sized such that the pressure drops during discharge are fully taken into account so that the vessel pressure is not excessive and also that the valve does not reseat instantly, i.e. chatter.

The maximum pressure drop of the pipework to the pressure relief valve should not exceed that specified in prEN 13648-3.

4.2.6.3 Outer jacket

A pressure relief device shall be fitted to the outer jacket. The device shall be set to open at a pressure which prevents collapse of the inner vessel and is not more than 0,5 bar. The discharge area of the pressure relief device shall be not less than 0,34 mm²/l capacity of the inner vessel and in any case need not exceed 5 000 mm².

4.2.6.4 Piping

Any section of pipework containing cryogenic fluid which can be isolated shall be protected by a relief valve or other suitable relief device.

4.2.7 Valves

4.2.7.1 General

Valves shall conform to EN 1626.

4.2.7.2 Isolating valves

To prevent any large spillage of liquid, a secondary means of isolation shall be provided for those lines emanating from below the liquid level that are:

- greater than 9 mm bore and exhausting to atmosphere; or
- greater than 50 mm bore when forming part of a closed system.

The secondary means of isolation may be within the user installation and shall provide an equivalent level of protection.

The secondary means of isolation, where provided, may be achieved, for example, by the installation of a second valve, positioned so that it can be operated safely in emergency, an automatic fail-closed valve or a non-return valve or fixed or removable cap on the open end of the pipe.

4.2.8 Filling ratio

Means shall be provided to ensure that the vessel is not filled to more than 98 % of its total volume with liquid at the filling condition.

4.3 Design by calculation

4.3.1 General

When design is by calculation in accordance with 4.1.2, the dimensions of the inner vessel and outer jacket shall not be less than that determined in accordance with this subclause.

4.3.2 Inner vessel

4.3.2.1 General

The information in 4.3.2.2 to 4.3.2.6 shall be used to determine the pressure part thicknesses in conjunction with the calculation formulae of 4.3.6.

4.3.2.2 Design loads and allowable stresses

- a) In accordance with 4.2.3.2 j) 1)

Material properties determined either in accordance with 4.3.2.3.2 or 4.3.2.3.3 shall be adopted at the discretion of the vessel manufacturer.

- b) In accordance with 4.2.3.2 j) 2), 3), 4) and 5)

Material properties determined in accordance with 4.3.2.3.2 shall be adopted.

4.3.2.3 Material property K

4.3.2.3.1 General

The material property K to be used in the calculations shall be as follows:

- for austenitic stainless steel and unalloyed aluminium, 1 % proof strength;
- for all other metals the yield strength, and if not available 0,2 % proof strength.

NOTE Upper yield strength can be used.

4.3.2.3.2 K_{20}

K shall be the minimum value at 20 °C taken from the material standard, (see annex J).

4.3.2.3.3 K_t

The permissible value of K shall be determined for the material at the operating temperature corresponding to the saturation temperature at the maximum allowable pressure of the vessel, of the contained cryogenic fluid. The value of K and E shall be determined from the material standard (see EN 10028-7:2000, annex F for austenitic stainless steels) or shall be guaranteed by the material manufacturer.

4.3.2.4 Safety factors S , S_T , S_p and S_k

Safety factors are the ratio of material property K over the maximum allowable stress.

a) internal pressure (pressure on the concave surface):

- at vessel maximum allowable pressure

$$S = 1,5$$

- at vessel proof test pressure

$$S_T = 1,05$$

b) external pressure (pressure on the convex surface):

- cylinders and cones $S_p = 1,6$

$$S_k = 3,0$$

- spherical region $S_p = 2,4$

$$S_k = 3,0 + 0,002 R/s$$

- knuckle region $S_p = 1,8$

4.3.2.5 Weld joint factors v

For internal pressure (pressure on the concave surface) $v = 0,85$ or $1,0$ (see clause 6, Table 6).

For external pressure (pressure on the convex surface) $v = 1,0$.

4.3.2.6 Allowances c

$$c = 0$$

4.3.3 Outer jacket

4.3.3.1 General

The following shall be used to determine the pressure part thicknesses in conjunction with the calculation formulae of 4.3.6.

4.3.3.2 Design pressure p

The internal design pressure p shall be equal to the set pressure of the outer jacket pressure relief device.

The external design pressure p shall be 1 bar.

4.3.3.3 Material property K

The material property K to be used in the calculations shall be at 20 °C, as defined in 4.3.2.3.

4.3.3.4 Safety factors S , S_p and S_k

Internal pressure (pressure on the concave surface)

$$S = 1,1$$

External pressure (pressure on the convex surface)

— cylinders and cones $S_p = 1,1$

$$S_k = 2,0^{*)}$$

— spherical region $S_p = 1,6$

$$S_k = 2,0 + 0,001\,4\,R/s$$

— knuckle region $S_p = 1,2$

4.3.3.5 Weld joint factors v

For internal pressure (pressure on the concave surface) $v = 0,7$.

For external pressure (pressure on the convex surface) $v = 1,0$.

*) For well proven designs a factor of safety S_k equal to 1,5 is acceptable provided that :

D is not more than 2 300 mm ;

l_b is not more than 10 200 mm ;

and the annular space is perlite insulated.

4.3.3.6 Allowances c

No allowance is required.

$$c = 0$$

NOTE External surfaces should be adequately protected against corrosion.

4.3.4 Supports and lifting points

The supports and lifting points shall be designed for the loads defined in 4.2, using established structural design methods and safety factors.

When designing the inner vessel support system the temperature and corresponding mechanical properties to be used may be those of the component in question when the inner vessel is filled to capacity with cryogenic fluid.

4.3.5 Piping and accessories

Piping shall be designed for the loads defined in 4.2.3.7 using established piping design methods and safety factors.

4.3.6 Calculation formulae

4.3.6.1 Cylinders and spheres subject to internal pressure (pressure on the concave surface)

4.3.6.1.1 Field of application

Cylinders and spheres where:

$$D_a/D_i \leq 1,2$$

4.3.6.1.2 Openings

For reinforcement of openings see 4.3.6.7.

4.3.6.1.3 Calculation

The required minimum wall thickness s is:

— for cylinders

$$s = \frac{D_a p}{20(K/S)v + p} + c \quad (1)$$

— for spheres

$$s = \frac{D_a p}{40(K/S)v + p} + c \quad (2)$$

4.3.6.2 Cylinders subject to external pressure (pressure on the convex surface)

4.3.6.2.1 Field of application

Cylinders where

$$D_a/D_i \leq 1,2$$

4.3.6.2.2 Openings

Openings shall be calculated in accordance with 4.3.6.7 using for the pressure in the formula a value equal to the external pressure as though it were internally applied.

4.3.6.2.3 Calculation

Calculations shall be performed for elastic buckling and for plastic deformation. The lowest calculated pressure p_e or p_p shall not be less than the external design pressure.

NOTE 1 The buckling length l_b is the length of the unsupported cylinder (see Figures 1 and 2). Other forms of stiffening section can be used.

NOTE 2 For vessels with dished ends, the buckling length starts at the junction of the cylinder and the knuckle region of the dished end (see Figure 3).

4.3.6.2.4 Elastic buckling

Calculations are performed using the following formula:

$$p_e = \frac{E}{S_k} \left\{ \frac{20}{(n^2 - 1) \cdot \left[1 + (n/Z)^2 \right]^2} \cdot \frac{s - c}{D_a} + \frac{80}{12 \cdot (1 - \nu^2)} \cdot \left[n^2 - 1 + \frac{2n^2 - 1 - \nu}{1 + (n/Z)^2} \right] \cdot \left[\frac{s - c}{D_a} \right]^3 \right\} \quad (3)$$

where $Z = 0,5\pi D_a/l_b$ and n is an integer equal to or greater than 2 and greater than Z , so determined that the value for p_e is a minimum. n denotes the number of lobes produced by the buckling process which can occur at the circumference in the event of failure. The number of lobes can be estimated using the following approximation equation :

$$n = 1,63 \cdot \sqrt[4]{\frac{D_a^3}{l_b^2 (s - c)}} \quad (4)$$

For tubes and pipes, calculations may be performed using the following simplified formula:

$$p_e = \frac{E}{S_k} \cdot \frac{20}{(1 - \nu^2)} \cdot \left(\frac{s - c}{D_a} \right)^3 \quad (5)$$

If a test pressure higher than 1,25 p is specified, an additional assessment shall be made to ensure that the adopted material thickness is not less than that determined at test pressure with a safety factor of 0,74 S_k .

4.3.6.2.5 Plastic deformation

When $D_a/l_b \leq 5$

$$p_p = \frac{20K}{S_p} \cdot \frac{s-c}{D_a} \cdot \frac{1}{1 + \frac{1,5u(1-0,2D_a/l_b)D_a}{100(s-c)}} \quad (6)$$

When $D_a/l_b > 5$

the higher pressure obtained using equations (7) and (8) shall not be less than the external design pressure.

$$p_p = \frac{20K}{S_p} \cdot \frac{s-c}{D_a} \quad (7)$$

$$p_p = \frac{30K}{S_p} \left(\frac{s-c}{l_b} \right)^2 \quad (8)$$

If a test pressure higher than $1,25 p$ is specified, an additional assessment shall be made to ensure that the adopted material thickness is not less than that determined at the test pressure with a safety factor of $0,74 S_k$.

4.3.6.2.6 Stiffening rings

In addition to the ends, stiffening may be provided in the form of the examples shown in Figures 1 and 2. Other forms of stiffening may be used. The stiffening rings welded to the shell shall satisfy the following conditions:

$$I \geq \frac{0,124 p D_a^3}{10E} \sqrt{D_a(s-c)} \quad (9)$$

$$A \geq \frac{0,75 p D_a}{10K} \sqrt{D_a(s-c)} \quad (10)$$

The moment of inertia I is relative to the neutral axis of the reinforcing elements cross-section parallel to the shell axis (see axis $x - x$ in Figures 1 and 2). Narrow and high reinforcing elements of the kind shown in Figure 1 may undergo severe buckling. Where the height of the element is greater than 8 times its width, a more accurate calculation shall be made.

Where stiffening rings are joined to the shell by means of intermittent welds, the fillet welds at each side shall cover at least one third of the shell circumference and the number of weld discontinuities shall be at least $2n$. The number of buckling lobes n is obtained as indicated in 4.3.6.2.4.

If a test pressure, p_T , higher than $1,25 p$ is specified, an additional assessment shall be made to ensure that the adopted values of I and A are not less than those determined by formulae (9) and (10) at a pressure of $0,74 p_T$.

4.3.6.3 Spheres subject to external pressure (pressure on the convex surface)

Spheres subject to external pressure shall be evaluated in accordance with the appropriate part of 4.3.6.4.4.

4.3.6.4 Dished ends subject to internal or external pressure

4.3.6.4.1 Field of application

Hemispherical ends where $D_a/D_i \leq 1,2$

10 % torispherical ends where $R = D_a$ and $r = 0,1 D_a$

and

2:1 torispherical ends where $R = 0,8 D_a$ and $r = 0,154 D_a$

In the case of torispherical ends $0,001 \leq \frac{(s-c)}{D_a} \leq 0,1$

NOTE Other end shapes can be used provided suitable calculations are carried out.

4.3.6.4.2 Straight flange

The straight flange length h_1 (Figure 4a)) shall be not less than:

- for 10 % torispherical ends, 3,5 s ;
- for 2:1 torispherical ends, 3,0 s .

The straight flange may be shorter providing that in the case of inner vessels the circumferential joint between the dished end and the cylinder is non-destructively tested as required for a weld joint factor of 1,0.

NOTE Other flange/weld configurations can be used providing suitable calculations are carried out.

4.3.6.4.3 Intermediate heads

Heads, without limit to thickness, may be installed in accordance with of Figure F.2. The outside diameter of the head skirt shall be a close fit inside the ends of the adjacent sections of the cylinder.

The butt weld and fillet weld shall be adequately sized to jointly resist any relevant pressure, mechanical and thermal loads. This may be achieved by accurate detailed stress analysis and by adopting the criteria for acceptable stresses of annex A.

Where only pressure stresses are present, a simplified approach may be adopted such that the butt weld and fillet weld are sized to resist in shear a load equivalent to 1,5 times the maximum differential pressure across the head multiplied by the cross sectional area of the shell.

The allowable shear stress in this simplified case should not exceed $K/3$ where the area of the butt weld in shear is the width at the root of the weld multiplied by the circumferential length of the weld and the area of the fillet weld is the throat thickness multiplied by times the circumferential length of the weld.

Where the stresses in the attachment are fully analysed and assessed in accordance with annex A, the fillet weld may be omitted. In other cases the fillet weld shall be continuous.

4.3.6.4.4 Internal pressure calculation (pressure on concave surface)

4.3.6.4.4.1 Crown and hemisphere thickness

The wall thickness of the crown region of dished ends and of hemispherical ends shall be determined using 4.3.6.1.3 for spheres with $D_a = 2(R + s)$.

Openings within the crown area of 0,6 D_a of torispherical ends, see Figure 4b), and in hemispherical ends shall be reinforced in accordance with 4.3.6.7. When pad type reinforcement is used the edge of the pad shall not extend beyond the area of 0,8 D_a for 10 % torispherical ends or 0,7 D_a for 2:1 torispherical ends.

4.3.6.4.4.2 Torispherical end knuckle thickness and hemispherical end to shell junction thickness

The required thickness of the knuckle region and hemispherical end junction shall be:

$$s = \frac{D_a p \beta}{40 \left(\frac{K}{S} \right) v} + c \quad (11)$$

For hemispherical ends a β value of 1,1 shall be applied within the distance x from the tangent line joining the end to the cylinder,

where

$$x = 0,5 \sqrt{R(s - c)}$$

β is taken from Figure 5 for 10 % torispherical ends and from Figure 6 for 2:1 torispherical ends as a function of $\frac{(s - c)}{D_a}$. Iteration is necessary.

When there are openings outside the area 0,6 D_a the required thickness is found using β from Figures 5 and 6 using the appropriate curve for the relevant value of $\frac{d_i}{D_a}$.

The β factor is derived from the lower curves of Figures 5 and 6 when there are no openings outside the area 0,6 D_a .

D_a is the diameter of the end as shown in Figures 4a) and 4b).

4.3.6.4.4.3 If a domed end is welded together from crown and knuckle components, the joint shall be at a sufficient distance x from the knuckle. The distance regarded as sufficient is as follows, but with a minimum, however, of at least 100 mm (see Figure 4c)):

— the crown and knuckle are of different wall thickness:

$$x = 0,5 \sqrt{R(s - c)}$$

where s is the required wall thickness of the knuckle.

— the crown and knuckle are of equal wall thickness:

— for 10 % torispherical ends $x = 3,5 s$;

— for 2:1 torispherical ends $x = 3,0 s$.

$\nu = 1,0$ may be applied if the scope of testing corresponds to that specified for a design stress level equal to the permissible design stress level or in the case of one-piece ends.

$\nu = 1,0$ may also be applied in the case of welded domed ends - except hemispherical ends - regardless of the scope of testing provided the weld intersects the crown area of $0,6 D_a$ (see Figures 4e) and 4f (left-hand side)).

4.3.6.4.4 If the ligament on the connecting line between adjacent openings is not entirely within the $0,6 D_a$ region the ligament shall not be less than half the sum of the opening diameters.

4.3.6.4.5 External pressure calculations (pressure on the convex surface)

Elastic buckling

There is adequate resistance to elastic buckling when:

$$p \leq 3,66 \frac{E}{S_k} \left(\frac{s-c}{R} \right)^2 \quad (12)$$

If a test pressure higher than $1,25 p$ is specified, the adopted material thickness shall not be less than:

$$s = R \cdot \sqrt{\frac{0,2 \cdot S_k \cdot \text{Test pressure}}{E}} + c \quad (13)$$

where

S_K safety factor determined from 4.3.2.4 or 4.3.3.4.

Plastic deformation and plastic buckling

Resistance to plastic deformation shall be determined by using 4.3.6.4.4 with the appropriate safety factor S_p defined in 4.3.2.4 and 4.3.3.4.

If a test pressure higher than $1,25 p$ is specified, an additional assessment shall be made to ensure that the adopted material thickness is not less than that determined at the pressure with a safety factor not less than $0,74 S_k$.

4.3.6.5 Cones subject to internal or external pressure

4.3.6.5.1 Symbols and units

For the purposes of 4.3.5.5, the following symbols apply in addition to those given in 3.2:

A	area of reinforcing ring	mm ²
D_{a1}	outside diameter of connected cylinder (see Figure 7)	mm
D_{a2}	outside diameter at effective stiffening (see Figure 9)	mm
D_k	design diameter (see Figure 7)	mm
D_s	shell diameter at nozzle (see Figure 8)	mm
I	moment of inertia about the axis parallel to the shell	mm ⁴
l	cone length between effective stiffenings (see Figure 9)	mm

s_g	required wall thickness outside corner area	mm
s_l	required wall thickness within corner area	mm
x_i	characteristic lengths ($i = 1,2,3$) to define corner area (Figures 7a and 7b and 4.3.5.5.5)	mm
φ	cone angle	°
r	inside radius of knuckle	mm

4.3.6.5.2 Field of application

Cones according to Figure 7 where:

$$0,001 \leq \frac{s_g^c}{D_{a1}} \leq 0,1$$

and

$$0,001 \leq \frac{s_1^c}{D_{a1}} \leq 0,1$$

Small ends with a knuckle can be safely assessed and verified as a small end with a corner joint.

For external pressure $|\varphi| \leq 70^\circ$

Other cone angles may be used providing suitable calculations are carried out.

4.3.6.5.3 Openings

Openings outside of the corner area (see Figure 8) shall be designed as follows.

If $|\varphi| < 70^\circ$ design according to 4.3.6.7 using an equivalent cylinder diameter of:

$$D_i = \frac{D_s + d_i |\sin \varphi|}{\cos \varphi} \quad (14)$$

$|\varphi| \geq 70^\circ$ design according to 4.3.5.6.

4.3.6.5.4 Non destructive testing

All corner joints shall be subject to the examination required for a weld joint factor v of 1,0. See Table 6 in clause 6.

4.3.6.5.5 Corner area

The corner area is that part of the cone where the dominant stresses are bending stresses in the longitudinal direction.

The corner area is defined in Figures 7a and 7b by x_1 , x_2 , x_3 calculated from the following equations:

$$x_1 = \sqrt{D_{a1}(s_l - c)} \quad (15)$$

$$x_2 = 0,7 \sqrt{\frac{D_{a1}(s_l - c)}{\cos \varphi}} \quad (16)$$

$$x_3 = 0,5x_1 \quad (17)$$

4.3.6.5.6 Internal pressure calculation (pressure on concave surface) $|\varphi| \leq 70^\circ$

a) within corner area

The required wall thickness (s_1) within the corner area is calculated from Figures 10 a) to g) for the large end and Figure 10 h) for the small end of a cone using the following variables:

$$\varphi, \frac{pS}{15KV}, \text{ and } \frac{r}{D_{a1}}$$

For a corner joint use the curve for $\frac{r}{D_{a1}} = 0$.

For intermediate cone angles use linear interpolation. The wall thickness s_l in the corner area shall not be less than the required thickness s_g outside of the corner area as calculated in formula (18).

b) outside corner area

The required wall thickness, s_g , outside the corner area is calculated from:

$$s_g = \frac{D_k p}{20 \frac{K}{S} v - p} \div \frac{1}{\cos \varphi} + c \quad (18)$$

where

for the large end, $D_k = D_{a1} - 2[s_1 + r(1 - \cos \varphi) + x_2 \sin \varphi]$.

For the small end, D_k is the maximum diameter of the cone, where the wall thickness is s_g .

4.3.6.5.7 Internal pressure calculation (pressure on the concave surface) $|\varphi| > 70^\circ$

If $r \geq 0,01 D_{a1}$ the required wall thickness is

$$s_l = s_g = 0,3(D_a l - r) \frac{|\varphi|}{90} \sqrt{\frac{p}{10 \left(\frac{k}{S}\right)^v}} + c \quad (19)$$

4.3.6.5.8 External pressure calculation (pressure on the convex surface)

Stability against elastic buckling and plastic deformation shall be verified using 4.3.6.2 and an equivalent cylinder.

For the example shown in Figure 9 the equivalent cylinder diameter between the knuckle and the stiffener is:

$$D_a = \frac{D_{a1} + D_{a2}}{2 \cos|\varphi|} \quad (20)$$

and the equivalent cylinder length is:

$$l = \frac{D_{a1} - D_{a2}}{2 \sin|\varphi|} \quad (21)$$

Depending on the relevant boundary conditions the equivalent length between two effective stiffening sections shall be reliably estimated within the meaning of 4.3.6.2.

When $\varphi \geq 10^\circ$ the corner area of a large end can be considered as effective stiffening.

For small ends the thickness in the corner area shall not be less than 2,5 times the required thickness of the conical shell with the same angle $|\varphi|$ or a stiffener shall be fitted with the following properties:

$$l \geq \frac{p(D_{a1})^4}{960 \left(\frac{E}{S_k} \right)} \tan|\varphi| \quad (22)$$

If a test pressure higher than $1,25 p$ is specified, an additional assessment shall be made to ensure that the adopted value of l is not less than that determined at the test pressure with a safety factor of $0,74 S_k$.

$$A \geq \frac{p(D_{a1})^2}{80 \left(\frac{K}{S_p} \right)} \tan|\varphi| \quad (23)$$

If a test pressure higher than $1,25 p$ is specified, an additional assessment shall be made to ensure that the adopted value of A is not less than that determined at the test pressure with a safety factor of $0,74 S_p$.

S_k (cylinder) is the safety factor to prevent elastic buckling from 4.3.2.4 or 4.3.3.4.

S_p (cylinder) is the safety factor to prevent plastic deformation from 4.3.2.4 or 4.3.3.4.

D_{a1} is the diameter according to Figure 7b).

The shell over a width of $0,5\sqrt{D_{a1}s_1}$ can be used to calculate the moment of inertia and the area.

In addition the corner joint should not be regarded as a classical boundary condition i.e. the overall length should be formed from the individual meridional length of the cone and cylinder.

In addition, the cone shall be verified using 4.3.6.5.6 and the safety factors S_p for cylinders from 4.3.2.4 or 4.3.3.4.

If a test pressure higher than $1,25 p$ is specified, an additional assessment shall be made to ensure that the adopted material thickness is not less than that determined at the test pressure with a safety factor of $0,74 S_k$. For thickness calculations in the corner area v shall be the value applicable for internal pressure.

4.3.6.6 Flat ends

4.3.6.6.1 Symbols

For the purposes of 4.3.6.6, the following symbols apply in addition to those given in 3.2:

- d_1, d_2 etc. opening diameters in mm;
- D_1, D_2 etc. flat end diameters in mm.

4.3.6.6.2 Field of application

Welded or solid flat ends where Poisson's ratio is approximately 0,3, and

$$\frac{(s-c)}{D} \geq 4 \sqrt{\frac{0,0087 p}{E}}$$

and

$$\frac{(s-c)}{D} \leq \frac{1}{3}$$

4.3.6.6.3 Openings

Openings are calculated in accordance with 4.3.6.6.4 but with the C factor multiplied by C_A , where C_A is given in Figure 11.

4.3.6.6.4 Calculation

The required minimum wall thickness of a circular flat end is:

$$s = CD_1 \sqrt{\frac{0,1pS}{K}} + c \quad (24)$$

C and D_1 are taken from Figure 12.

The required minimum wall thickness of a rectangular or elliptical flat end is

$$s = CC_E f \sqrt{\frac{0,1pS}{K}} + c \quad (25)$$

where C_E is taken from Figure 13.

4.3.6.7 Openings in cylinders, spheres and cones

4.3.6.7.1 Symbols and units

For the purposes of 4.3.6.7, the following symbols apply in addition to those given in 3.2:

b	width of pad, ring or shell reinforcement	mm
h	thickness of pad-reinforcement	mm
l	ligament (web) between two nozzles	mm

l'_s	length of nozzle reinforcement outstandings	mm
m	protruding length of nozzle	mm
s	length of nozzle reinforcement instand	mm
s_A	required wall thickness at opening edge	mm
s_S	wall thickness of nozzle	mm
t	In this context : centre-to-centre distance between two nozzles	mm

4.3.6.7.2 Field of application

Round openings and the reinforcement of round openings in cylinders, spheres and cones within the following limits:

$$0,002 \leq \frac{(s-c)}{D_a} \leq 0,1$$

$$\frac{(s-c)}{D_a} < 0,002 \text{ is acceptable if } \frac{d_i}{D_a} \leq \frac{1}{3}.$$

These rules only apply to cones if the wall thickness is determined by the circumferential stress.

NOTE 1 Additional external forces and moments are not covered by this sub-clause and are to be considered separately where necessary.

NOTE 2 These design rules permit plastic deformations of up to 1 % at highly stressed local areas during pressure test. Openings should therefore be carefully designed to avoid abrupt changes in geometry.

The design rules for non perpendicular nozzles shall be based on a perpendicular nozzle, using the dimension of the major elliptical axis or shall be calculated in accordance with EN 13445-3.

4.3.6.7.3 Reinforcement methods

Openings may be reinforced by one or more of the following typical but not exclusive methods:

- increase of shell thickness, see Figures 14 and 15;
- set in or set on ring reinforcement, see Figures 16 and 17;
- pad reinforcement, see Figure 18;
- increase of nozzle thickness, see Figures 19 and 20;
- pad and nozzle reinforcement, see Figure 21.

Where ring or pad reinforcement is used on the inner vessel the space between the two fillet welds shall be vented into the vacuum interspace.

4.3.6.7.4 Design of openings

The fillet weld on a reinforcing pad shall have a minimum throat thickness of half of the pad thickness.

The critical dimensions of each nozzle to shell weld shall be not less than the required thickness of the thinner part.

Where the strength of the reinforcing material is lower than the strength of the shell material an allowance in accordance with 4.3.6.7.5 shall be made in the design calculations. If the strength of the reinforcing material is higher than the strength of the shell material no allowance for the increased strength is permitted.

4.3.6.7.5 Calculation

Where the material property K of the reinforcement is lower than that of the shell the cross section of pad reinforcement and the thickness of nozzle reinforcement shall be reduced by the ratio of K values. In the case of a shell subjected only to internal pressure, with a row of nozzles joined to the shell by fully penetrating welds, it is not necessary to calculate the individual reinforcement required for each nozzle. However the thickness of the shell to resist internal pressure shall be calculated using the least value of weakening factor of either v_A obtained from equation (34) or v .

Openings shall also be reinforced according to the following relationship:

$$\frac{p}{10} \left(\frac{A_p}{A_\sigma} + \frac{1}{2} \right) \leq \frac{K}{S} \quad (26)$$

which is based on equilibrium between the pressurised area A_p and the load bearing cross sectional area A_σ . The wall thickness obtained from this relationship shall be not less than the thickness of the unpierced shell.

The pressurised area A_p and the load bearing cross sectional area A_σ which equals $A_{\sigma 0} + A_{\sigma 1} + A_{\sigma 2}$ are obtained from Figures 22 to 25.

The maximum extent of the load bearing cross sectional area shall be not more than b as defined in formula (28) for shells and l_s as defined in formulae (30) or (31) for nozzles, as appropriate.

The protrusion of nozzles l'_s may be included as load bearing cross sectional area up to a maximum length of:

$$l'_s = 0,5 l_s$$

The restrictions of 4.3.6.7.7 and 4.3.6.7.8 shall be observed.

If the material property K_1 , K_2 etc. of the reinforcing material is lower than that of the shell the dimensions shall comply with:

$$\left(\frac{K}{S} - \frac{p}{20} \right) A_{\sigma 0} + \left(\frac{K_1}{S} - \frac{p}{20} \right) A_{\sigma 1} + \left(\frac{K_2}{S} - \frac{p}{20} \right) A_{\sigma 2} \geq \frac{p}{10} A_p \quad (27)$$

4.3.6.7.6 Ring or pad reinforcement or increased shell thickness

If the actual wall thickness of the cylinder or sphere is less than the required thickness s_A at the opening, the opening is adequately reinforced if the wall thickness s_A is available round the opening over a width of:

$$b = \sqrt{(D_i + s_A - c)(s_A - c)} \quad (28)$$

with a minimum of $3 s_A$ (see Figures 16, 17 and 18).

For calculation purposes s_A shall be limited to not more than twice the actual wall thickness.

The thickness of pad reinforcement in accordance with Figure 18 preferably shall be not more than the actual wall thickness to which the pad is attached.

Internal pad reinforcement is not allowed.

The width of the pad reinforcement may be reduced to b_1 provided the pad thickness is increased to h_1 according to:

$$b_1 \cdot h_1 \geq b \cdot h \quad (29)$$

and the limits given above are observed.

4.3.6.7.7 Reinforcement by increased nozzle thickness

For calculation purposes s_s shall be not more than twice the actual wall thickness.

The thickness of the nozzle shall preferably be not greater than twice the actual shell thickness.

The wall thickness s_A at the opening shall extend over a width b in accordance with formula (28) with a minimum of $3 s_A$.

The limits of reinforcement normal to the vessel wall are:

— for cylinders and cones, $l_s = 1,25 \sqrt{(d_i - s_s - c)(s_s - c)}$ (30)

— for spheres, $l_s = \sqrt{(d_i - s_s - c)(s_s - c)}$ (31)

The length l_s may be reduced to l_{s1} provided that the thickness s_s is increased to s_{s1} according to the following:

$$l_{s1} \cdot s_{s1} \geq l_s \cdot s_s \quad (32)$$

and the limits given above are observed.

4.3.6.7.8 Reinforcement by a combination of increased shell and nozzle thicknesses

Shell and nozzle thicknesses may be increased in combination for the reinforcement of openings (see Figure 21). For the calculation of reinforcement 4.3.6.7.6 and 4.3.6.7.7 shall be applied together. The increase in shell thickness may be achieved by an actual increase in shell thickness or the addition of a pad.

4.3.6.7.9 Multiple openings

Multiple openings are regarded as single openings provided the distance l between two adjacent openings, Figures 24 and 26, complies with:

$$l \geq 2 \sqrt{(D_i + s_A - c)(s_A - c)} \quad (33)$$

If l is less than required by formula (33) a check shall be made to determine whether the cross section between openings is able to withstand the load acting on it. Adequate reinforcement is available if the requirement of formula (26) or (27), as appropriate is met.

Where adjacent openings in a cylinder are arranged intermediately between the longitudinal and circumferential direction the calculation scheme for the longitudinal direction (see Figure 24) shall be applied, but the part of the pressurised area corresponding to the unpierced cylinder $\left(\frac{tD_i}{2}\right)$ may be reduced with an arrangement factor $= 0,5 (1 + \cos^2 \varphi)$.

See Figure 25 for angle φ .

Nozzles joined to the shell in line by full penetration welds with the wall thickness calculated for internal pressure only may be designed with a weakening factor:

$$v_A = \frac{(t - d_i)}{t} \quad (34)$$

If the nozzles are not attached by full penetration welds, D_a shall be used in formula (34).

4.3.7 Design by analysis

Unless the design has been validated by experiment, calculations in addition to those in 4.3.6 may be required to ensure that stresses due to operating loads are within acceptable limits. All load conditions expected during service shall be considered (see 4.2.3).

In these calculations static loads are substituted for static plus dynamic loads.

The analysis shall take account of gross structural discontinuities, but need not consider local stress concentrations.

Annex A provides terminology and acceptable stress limits when an elastic stress analysis is performed.

Acceptable calculation methods include:

- finite element ;
- finite difference ;
- boundary element ;
- recognised text books, published papers, codes and standards.

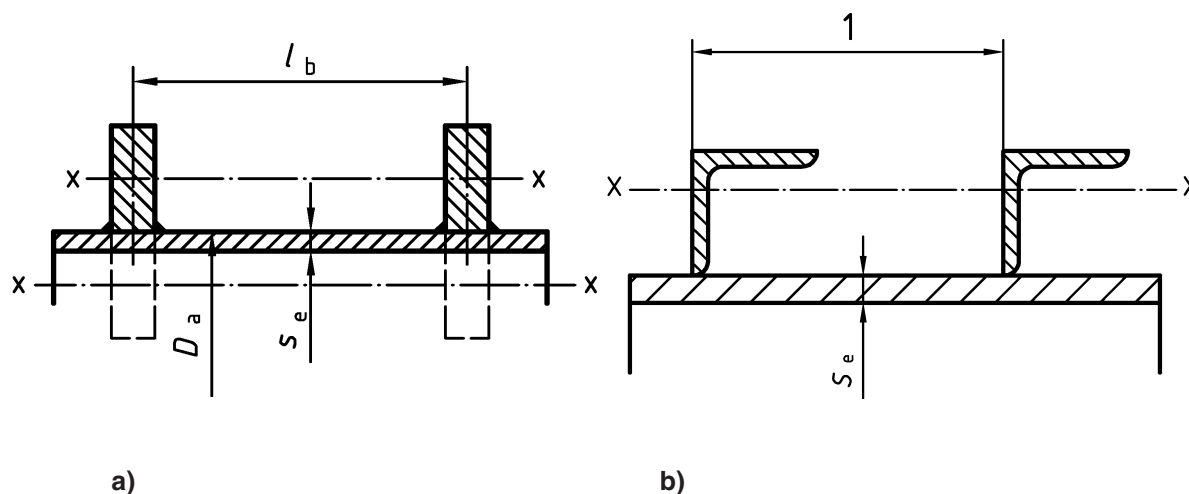


Figure 1 — Stiffening rings

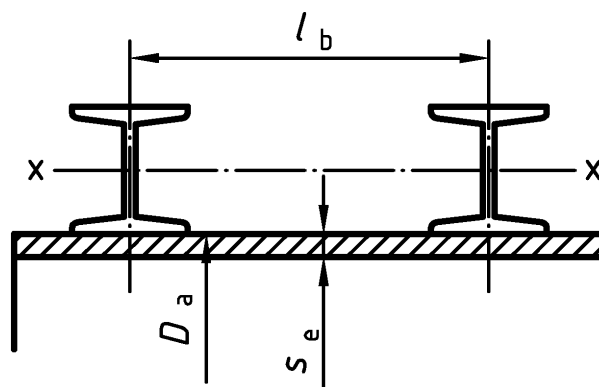


Figure 2 — Sectional materials stiffeners

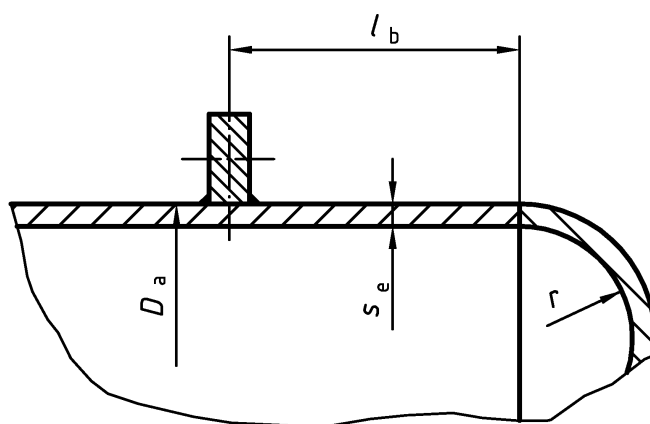


Figure 3 — Dished ends